EXPERIMENTAL INVESTIGATION OF THERMAL AND VENTILATION ANALYSIS FOR STRATUM VENTILATION – CFD STUDY

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Abstract- This investigates the air movement, air temperature profile and gaseous contaminant transportation in an individual specific room with stratum ventilation. Stratum ventilation has been proposed to cope for elevated indoor temperatures. The room temperature is elevated compared with conventional standards. Air speed, temperature and CO₂ concentration of a stratum ventilated room are investigated experimentally. The experimental investigation is carried out in an environmental chamber with the presence of heat generating components used to simulate an occupant (rectangles). The data obtained under well-defined conditions and therefore can be used for validating numerical models. Thermal comfort conditions and ventilation efficiency are studied based on the experimental results of four experimental cases. Thermal comfort indices are calculated from measured data. The values of these indices are found to satisfy the requirements of ASHRAE. For all the cases, the ventilation effectiveness is close to 1.5. This ventilation method could therefore be expected to provide indoor air quality in an efficient way. The agreements between the predicted values and experimental results are acceptable, which demonstrates the feasibility of simulating indoor airflows at elevated room temperature under stratum ventilation.

Keywords- Stratum Ventilation, Thermal Comfort, ASHRAE, IAQ, Case Study

I. INTRODUCTION

A. Ventilation
Ventilation moves outdoor air into a building or a room, and distributes the air within the building or room. The general purpose of ventilation in buildings is to provide healthy air for breathing by both diluting the pollutants originating in the building and removing the pollutants from it. There are three methods that may be used to ventilate a building: natural, mechanical and hybrid (mixed-mode) ventilation

II. THERMAL COMFORT

Thermal comfort analysis for a person standing in a room with an inlet and an outlet for air conditioning and a ceiling fan. Representative three dimensional (3D) simulation was also performed for the comparison of results obtained by using 2D and 3D models. Different cases in which the ceilings fan may be not in use or in use with different air speed normal to the plane of fan blades due to different rotational speeds were considered. Predicted mean vote was computed and used to assess the thermal comfort characteristics. It was found that as the normal air speed from the fan increases, thermal comfort significantly shifts toward the cooler scale to allow higher supply air temperature or higher heat load in the room while maintaining the same comfort level.

Budaiwi Ismail M. presented a systematic multi-phase and solution-oriented approach through which thermal-comfort problems can be assessed, identified and treated in a systematic way without utilizing unnecessary resources and time has been introduced. The approach can be helpful to building operators and facility managers when dealing with thermal-comfort problems. Pan Chung-Shu et al. [22] evaluated the performance of a personalized air conditioning system, namely an innovative partition-type fan-coil unit (PFCU), against that of a central air-conditioning system, in
terms of their thermal comfort provided and cooling energy consumed. For a cooling load given, it was found that the thermal comfort index (PMV) resulted from the personalized system is always lower than that from a central system. Also, the PMV-curve of the personalized system responds to the loads faster.

III. ASHRAE STANDARD 55

Thermal comfort is defined as the state of mind in which one acknowledges satisfaction with regard to the thermal environment. In terms of sensations, thermal comfort is described as a thermal sensation of being neither too warm nor too cold, defined by the following seven-point thermal sensation scale proposed by ASHRAE:

<table>
<thead>
<tr>
<th>-3</th>
<th>-2</th>
<th>-1</th>
<th>0</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cold</td>
<td>Cool</td>
<td>Slightly cool</td>
<td>Neutral</td>
<td>slightly warm</td>
<td>warm</td>
<td>hot</td>
</tr>
</tbody>
</table>

Table 1. ASHRAE Thermal Sensation Scale

A steady-state energy balance is a necessary condition for thermal comfort, but is not sufficient by itself to establish thermal comfort. Fanger calculated the heat losses for a comfortable person, experiencing a neutral sensation, with corresponding skin temperature (Tsk) and regulatory sweating. The calculated heat losses L are then compared with the metabolic rate M. If \( M = M \), the occupant feels comfortable. If \( M > M \), then this person feels cool; and if \( M < M \), then this person feels warm. Using the responses of 1396 persons during laboratory experiments at Kansas State University of the United States and Technical University of Denmark, Fanger related the comfort data to physiological variables. At a given level of metabolic activity M, and when the body is not far from thermal neutrality, the mean skin temperature tsk and sweat rate are the only physiological parameters influencing the heat balance. However, heat balance alone is not sufficient to establish thermal comfort [2]. Fanger reduced the relationships between metabolism and heat loss to a single equation, which assumed all sweat generated is evaporated, eliminating clothing permeation efficiency as a factor in the equation. This assumption is valid for normal indoor clothing worn in typical indoor environments with low or moderate activity levels. At higher activity levels (Mact > 3 met), where a significant amount of sweating occurs even at optimum comfort conditions, this assumption may limit accuracy. The reduced equation is slightly different from the heat transfer equations developed here. The radiant heat exchange is expressed in terms of the Stefan-Boltzmann law (instead of using \( h_r \)), and diffusion of water vapor through the skin is expressed as a diffusivity coefficient and a linear approximation for saturated vapor pressure evaluated at tsk. The combination of environmental and personal variables that produces a neutral sensation may be expressed as follows:

\[
M - W = 3.96 \times 10^{-8} f_{cl} \left[ \left( t_{cl} + 273 \right)^4 - \left( t_{a} + 273 \right)^4 \right] + f_{cl} h_c \left( t_{cl} - t_a \right) + 3.05 [5.73 - 0.007 (M - W) - p_a] + 0.42 \left[ (M - W) - 58.15 \right] + 0.0173 M (5.87 - p_a) + 0.0014 M (34 - t_a)
\]

Where,
- \( M \) = rate of metabolic heat production, W/m\(^2\)
- \( W \) = rate of mechanical work accomplished, W/m\(^2\)
- \( R_{cl} \) = thermal resistance of clothing in m\(^2\)-K/W
- \( h_c \) = convective heat transfer coefficient, W/(m\(^2\)-K)
- \( f_{cl} \) = clothing area factor \( A_{cl} / A_D \) dimensionless
- \( A_c \) = surface area of clothed body
- \( A_D \) = DuBois surface area of nude body
Pa = water vapor pressure in ambient air, kPa
\( t_a \) = air temperature, K
\( t_r \) = mean radiant temperature, K
\( t_{cl} \) = temperature of outer surface of the clothed body, K
\( R_{cl} = 0.155/\text{clo} \)
1.0 clo is equivalent to 0.155 m\(^2\)·K/W.
1 met = 58.1 W/m\(^2\)

IV. TESTING FACILITY AND EXPERIMENTATION

A. Identification of Room for Experimentation

Test room with a 1 TR stratum ventilation in Test chamber is selected as shown in below:

![Schematic of the test room](image1)

*Figure 1. Schematic of the test room*

The test room is the size of \( L \times H \times W = 15 \text{ feet} \times 10 \text{ feet} \times 7 \text{ feet} \) with a nearly cubical shape as shown in the Fig. 3.1. The room facilitates 1 TR Stratum ventilation and a exhaust on top of the entrance wall as shown in Fig1. Ceiling fan is installed

B. Experimentation

![Test room grid points](image2)

*Fig.2 Test room grid points*
Experiments were conducted in Test room different conditions. In test room 1, tests were carried out at four conditions, condition 1 where stratum ventilation is placed at the opposite to entrance wall of the room at 3-4.5 feet height from floor. The floor was divided into $7 \times 5 = 35$ points of 1 feet $\times$ 3 feet square (Fig 2). The grid points are designated as $x_{ij}$ such as 11, 12, 13, 14, 15, 25, 24, as per the grid position on the array shown in Fig.2. At each grid point the velocity and temperature were recorded along the Z axis at the distances 2 feet, 4 feet, 6 feet and 8 feet from the floor, in sequence.

The experiments were carried out in test room for following conditions:

<table>
<thead>
<tr>
<th>Condition No.</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Stratum Ventilation and Fan both were OFF (Natural)</td>
</tr>
<tr>
<td>2</td>
<td>Stratum Ventilation ON &amp; Fan were OFF</td>
</tr>
<tr>
<td>3</td>
<td>Stratum Ventilation OFF &amp; Fan were ON</td>
</tr>
<tr>
<td>4</td>
<td>Stratum Ventilation ON and Fan ON</td>
</tr>
</tbody>
</table>

Atmospheric temperature and humidity measured at the start and end of the measurement by using sling psychrometer. To check the effect of the ambient conditions experiments were conducted at different calendar days.

V. RESULTS AND DISCUSSIONS

A. Procedure Of CFD Analysis For Test Room

1) Create 3D Model of Test Room in Creo 3.0. Save above model in *.IGES Format for Importing into ANSYS Workbench Mesh Module. Import above Saved *.IGES File in ANSYS Workbench

2) Number of Nodes: 876186
   Number of Element: 246664
   Save above meshed model in *.CMDB Format for Importing into ANSYS CFX.
   Import above *.CMDB File in ANSYS CFX PRE.
3) Define Volume Fraction for oxygen and carbon dioxide

**Fig 5** Meshing and Import *.CMDB File in ANSYS CFX PRE

**Fig 6** Image for $O_2$ Volume Fraction

**Fig 7** Image for $CO_2$ Volume Fraction
VI. CONCLUSION AND FUTURE SCOPE

A simplified model for 2-phase momentum, heat and mass transfer in test room was established to predict airflow around, air and temperature. The model equations were solved and validated by means of experimental data from a pilot test room. An error of about 20% for velocity magnitude prediction for test room was achieved. The model was capable of predicting the cooling rate of the air. Discrepancies in the temperature prediction are due to local under-prediction of the air velocity caused by the k-3 turbulence model, the assumption of uniform initial temperature distribution inside room and ignoring gradients inside the individual products. The model shows a rather good trend of cooling rate and can be used to study the effects of different parameters in the design and operation of industrial test rooms. Compared with mixing ventilation and displacement ventilation, stratum ventilation derives its energy saving potential largely from the following two factors: a reduced ventilation load and increased coefficients of performance (COP) for chillers. The year-round energy saving can be found to be substantial up to 30% to 40% when compared with displacement ventilation and mixing ventilation respectively.

Proper room air flow distribution and thermal comfort are the important aspects of the air-conditioning. The primary objective of the air-conditioning is to create thermal comfort environment which is largely depends on the IAQ and room air flow pattern. Further, room air flow pattern in a conditioned space is situation dependent and cannot be generalized. In view of this fact the proposed project work involves the experimental study to visualize and analysis the room air flow pattern under different conditions. Further, thermal comfort is also quantified.

REFERENCES